DESIGN AND VERIFICATION OF MECHANISMS FOR A LARGE FOLDABLE ANTENNA

Hans Jurgen Luhmann*, Carl Christian Etzler*, and Rudolf Wagner*

ABSTRACT

The Synthetic Aperture Radar (SAR) Antenna onboard the ESA Remote Sensing Satellite (ERS-1) is a 10-m x 1-m planar array. It is folded into a dense package for launch and deployed in orbit. The resulting three antenna conditions, i.e., stowed, deploying, and deployed, pose different and in some cases conflicting requirements. Numerous mechanisms were developed to meet these requirements. This paper presents the most characteristic design requirements and constraints, their impact on the design, and the resulting features of the mechanisms.

INTRODUCTION

The SAR Antenna (see Fig. 1) consists of five panels of 2 m x 1 m each. Flat rectangular slotted waveguides form the radiating surface. They are made from Carbon Fiber Reinforced Plastics (CFRP) for reasons of thermal stability and low mass. Very few CFRP sandwich beams, as mechanical stiffeners, provide integrity and stiffness for the waveguide array. The resulting maximum panel thickness is 0.060 m.

The panels are connected in four axes by four pairs of Panel Hinge Assemblies (PHA). Each assembly includes a pair of ball bearings in titanium brackets and allows mutual panel rotation by at least 180 deg.

A Deployable Truss Structure (DTS) made from CFRP tubes provides structural depth as a basis for high surface accuracy and high mechanical stiffness at minimum antenna mass. The DTS incorporates the locking devices for the deployed stage and also transfers drive forces to the panels during the deployment (see Fig. 2).

Deployment is driven, in several phases, by a number of drive mechanisms. These include, for each of the two antenna wings: a leaf spring in the long foldable bar of the DTS; a leg spring assembly in the outer panel axis; a speed controlled dc motor near the inner panel axis, and attached to the rigidly mounted center panel.

In stowed configuration the panels are folded to a stack of 0.3-m thickness. The DTS folds completely between the panels with no extra space. A Hold-down and Release Mechanism (HRM) keeps the panels fastened during launch and releases upon telecommand. The HRM includes six pretensioned

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clamping levers on the longitudinal sides of the panel stack and a spring driven cable release system with pyrotechnic cutter.

The deployment sequence as depicted in Figure 2 is characterized by four events:

1. Release of antenna package by the HRM

2. Deployment phase 1 (spring driven) of first antenna wing and latching

3. Deployment phase 2 (motor driven) of first antenna wing and simultaneous deployment phase 1' of second wing and corresponding latchings

4. Deployment phase 2' of second wing and latching.

The driving requirements for the antenna structural and mechanisms design are summarized as follows:

- Dimensions of aperture 10 m x 1 m

- Stowed volume to be minimized
  -> 2.05 m x 1.1 m x 0.65 m including rigid mounting frame

- Mass to be minimized
  -> 85 kg including mechanisms, rigid mounting frame, thermal hardware

- Stiffness
  lowest eigenfrequency >50 Hz stowed, >4 Hz deployed

- Drive force margin
  release/deployment driving forces >3 x resisting forces

- Release and latching shocks severely constrained

- Surface accuracy < +3 mm maximum including manufacturing, deployment, thermal, and other effects.

The following sections address the technical problems and solutions in detail.

MECHANISMS FOR LAUNCH

Mechanism-structure interaction is the characteristic feature of the antenna in launch configuration driving the design of the HRM. Based on the requirements indicated above, the HRM has to provide secure locking for launch and allow unconstrained reliable release in orbit.

The large panel surface area, and their limited thickness, and thus low structural depth, made it difficult to meet the eigenfrequency requirement

without excessive mass penalty. Simple locking devices, say at the four panel corners, would put the stiffness requirement entirely to the panel structure and make heavy panels due to the thickness limits. Such heavy panels would pose a severe eigenfrequency problem and further penalties in terms of mass and complexity in the deployed configuration.

Yet another efficient fixation of the panel package was prohibited by electrical requirements: a locking device at the center point of the panel area, though most efficient structurally, would have imposed unacceptable disturbance on the electrical antenna characteristics, resulting from necessarily large brackets in the electrically most sensitive aperture area.

The HRM concept, as implemented, does the following:

o   Leaves the electrical aperture surface unaffected

o   Provides clamping at optimum stiffness, thus allowing minimum panel
    mass for favorable deployed properties

o   Presents the most mass-efficient design overall

o   Allocates all significant mechanism masses to locations on the fixed
    mounting frame, with no impact on the sensitive deployed frequency.

The central feature of the HRM system is six clamps, located at the four corners of the panel stack and in the center of the longitudinal edges (see Fig. 3). A "release system" of cables and pulleys connects the clamps to the "drive system" of two redundant spring drives. All details of the system are designed to meet the release force margin (drive factor >3) under worst case conditions, w.r.t. temperature and various uncertainties. In case one of the two drive springs should fail, the margin is still >1.5.

The design of the clamps is shown in Figure 4. The clamps are hinged to the fixed center panel and their hinged release heads act upon brackets in the outermost panel in the stack. The three panels in between are provided with solid brackets at the required locations so that the pretension force of 3000 N per clamp is transferred on a stiff path.

In the clamps' locking head, the locking function could be separated from the release function. Locking is provided by a four bar linkage system where a toggle level is employed in overcenter position as the locking element. In order to ensure safe locking in the presence of loads and vibrations during launch, the toggle lever is blocked by an additional pawl. A feature of the release system is a multifunctional release lever. It serves as a balancing element during launch so that rotary oscillation induced through the ropes is limited to small oscillations and, thus, do not affect the position of the toggle lever. This is achieved by preloading the release lever by two springs. The release lever also serves as the unlocking element. After initiation of HRM release by ignition of the pyrotechnics and activation of the spring drives, the release lever is rotated to its endstop. During that
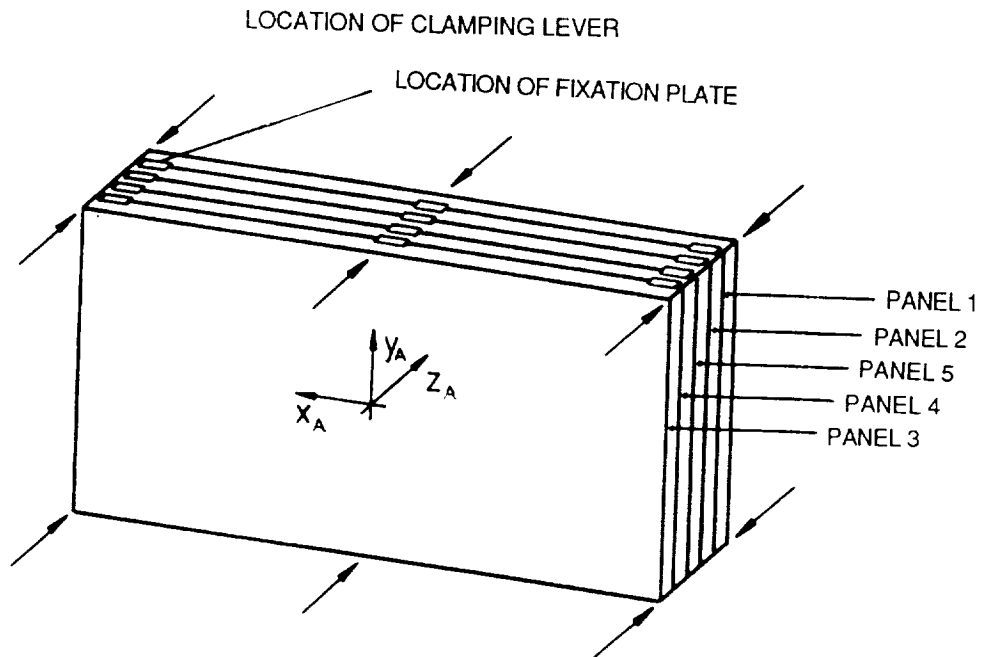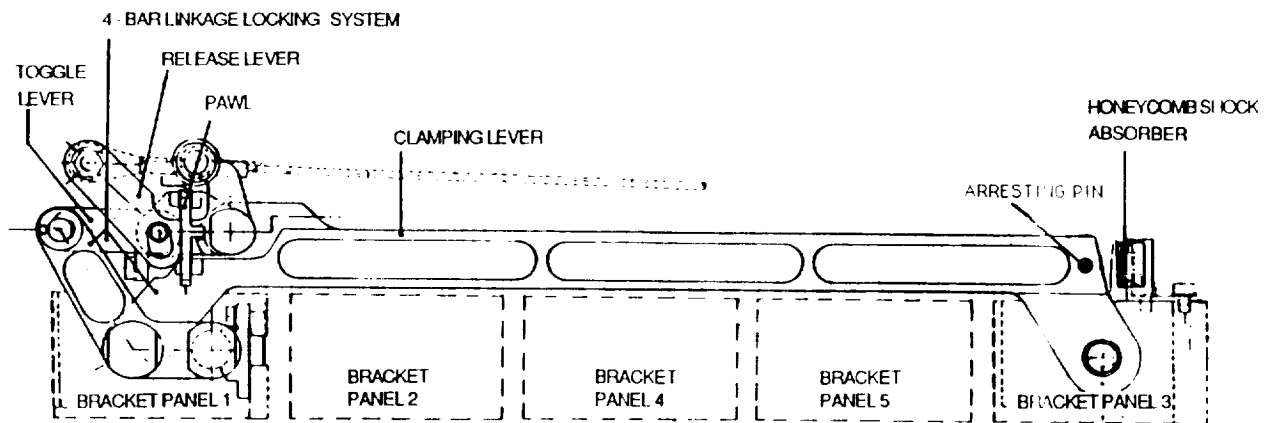
LOCATION OF CLAMPING LEVER

LOCATION OF FIXATION PLATE

PANEL 1
PANEL 2
PANEL 5
PANEL 4
PANEL 3

y_A
z_A
x_A

Figure 3. SAR-antenna launch configuration.

4 - BAR LINKAGE LOCKING SYSTEM

RELEASE LEVER

TOGGLE
LEVER
PAWL

CLAMPING LEVER

HONEYCOMB SHOCK
ABSORBER

ARRESTING PIN

BRACKET PANEL 1

BRACKET
PANEL 2

BRACKET
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Figure 4. HRM clamping lever.

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rotation the additional pawl is lifted off the endstop and the toggle lever is thus stretched and released.

To minimize shock loads, the rotating clamp is caught in a honeycomb absorber. It is then held in position by an arresting pin. The release system is designed such that three pairs of clamps are released in a sequence. Excessive release-force peaks are thus avoided, enabling a mass-efficient design of the spring drives.

A special development effort was required for the contact surfaces between the panels in the lines of action of the clamping forces. In order to achieve the stowed frequency requirement, those contact surfaces need to constrain all degrees of freedom between the panels, i.e., three displacements and three rotations. Upon release, they must in no way constrain the panel separation for reliable deployment. Detailed mechanical design and surface treatment of suitably-sized interface plates was the subject of a trade-off.

The selected surface design is shown in Figure 5. The plate, toothed in two orthogonal directions, constrains lateral displacements and normal rotation. Normal displacement and out-of-plane rotations are constrained by the preload force. The shaping as shown was selected for two reasons: the oblique surfaces of the teeth allow play-free contact between the two mating plates without posing exaggerated manufacturing tolerances for a perfect fit. One alternative considered employed two sets of in-plane shear pins, arranged in two orthogonal directions, in holes slightly excentric from the otherwise flat contact surface. Vibration tests on a development model showed unacceptable degradations resulting from manufacturing imperfections which could be avoided by an unacceptable effort only. The other advantage of the toothed shape is the distribution of the contact forces to a large number of smaller surfaces.

A concentration of pressure in a small area and possible welding or other surface damage is thus avoided. A flat surface design, with clamping along the edges of the rectangle only, was discarded since damage in the center of the pressure area was observed after vibration testing.

The area size of the interface plate and the amount of the preload force was determined by a dynamic response finite element analysis from the launch loads. The assumption of perfect clamping, which was needed for eigenfrequency analysis, requires that stresses between the plates remain of the pressure type, when superimposing the effects of pre-load and dynamic forces and moments. That assumption was confirmed valid for all interface planes between panels, except for the one nearest the fixed panels. The highest dynamic loads are encountered here. Since neither the pre-load, nor the interface plate size could be suitably increased, the "clamping" assumption was dropped for the six contact points in that plane and the analysis was re-run with "hinged" connections. Acceptable performance was thus found. The interface plate design at those locations was adapted to the hinge solution and satisfactory performance was confirmed in the antenna vibration and deployment testing.

MECHANISMS FOR DEPLOYMENT

Several deployment configurations for the planar antenna have been investigated prior to development. Due mainly to the limited space at launch, all configurations are based on a five-panel solution. In stowed position, all panels are folded to a stack including all supporting elements in the space between.

One candidate was an antenna structure deployed and supported with a pantograph truss. Because of the complex design and the large number of bars and hinged joints for the pantograph, which would result in alignment problems, this design was dropped. Another solution was characterized by a planar structure without a supporting truss. Detailed investigations of that configuration showed that the required in-orbit stiffness would be difficult to meet. During these studies the necessity of the supporting structure has been perceived in order to provide structural depth and thus to provide stiffness in the deployed configuration. On this basis, the selected configuration was developed: an antenna structure system with deployable truss structure and rigid mounted center panel. This solution was preferred because of the following:

- Moderate number of bars and hinged joints

- High stiffness for light-weight design

- Maximum base for alignment determined by rigid-mounted central panel.

A disadvantage of this design is the two wing deployment which required separate drive and locking mechanisms for each wing. The actual deployment sequence is shown in Figure 2. Each wing deployment is performed in two independent steps:

- Deployment and latching of the outermost panel (Phase 1)

- Deployment and latching of the two panel wing (Phase 2).

For the drive mechanisms, a combination of spring drives (Phase 1) and motor drives (Phase 2) was developed in consideration of the primary requirements given above.

Deployment Mechanisms Phase 1

For the deployment of the upper panel from the panel package (panel 1), a low-weight drive is required. The application of a motor drive would be problematical because of the comparatively high mass. This would require higher panel stiffness for launch,and especially when deployed. In consequence of this, the panel mass would increase and space problems would arise due to the thicker stiffeners.

A principle demand to the design was to balance the conflicting requirements of drive torque margin at all deployment configurations versus low shock loads at the end of deployment. It was a further design goal to develop a spring drive with approximately constant drive torque, corresponding to the nearly constant resistive torques versus deployment angle. A combination of two spring drives - leaf spring plus leg spring - was chosen. The leaf spring drive consists of two parallel C-shaped leaf springs and forms a part of the foldable bar of the DTS (see Fig. 2). Beside the low weight, the triple function of this unit is a remarkable feature. It combines the function of the following:

- A hinge (enables folding of the deployable truss structure for launch configuration)

- A drive (deployment drive during phase 1)

- A latching mechanism (performs arresting of panel 1 due to the high stiffness in stretched position).

The drive characteristic of the leaf spring is included in Figure 6. Due to the decreasing drive torque near the end of deployment, the torque margin requirement is not met. This deficiency could have been improved by thicker leaf springs, but could not be accomplished here, due to the limited yield point, and space between the panels. Therefore an additional spring drive was introduced, called a leg spring drive. This lightweight drive is located at the outer axis of the antenna (see Fig. 2). The spring elements are placed eccentrically to the panel axis of rotation. Based on this arrangement, it was possible to provide a special spring characteristic, like a sinus half-wave, with its maximum at the middle deployment position, balancing the insufficient behavior of the leaf spring drive. The superimposed spring characteristic is shown in Figure 6. The superposition led to an approximately constant drive torque during deployment. Increased levels still exist at the start of deployment, caused by compressive loads in the stowed configuration, and at the end of deployment, due to the stretching of the leaf springs.

Deployment Mechanisms Phase 2

The deployment of the 2-panel wings is performed by motor drives as depicted in Figure 2. Spring drives were also taken into consideration, but their features of low mass and simplicity are more than counterbalanced by a number of reasons. One problem was initialization of this deployment step at the required time within the overall deployment sequence. Another problem arose from high resistive torque which is caused by the simultaneous rotation of all PHA bearings and nearly all DTS bearings. Further, the resistive torque of the cable harness between the intermediate panel (e.g., panel 2) and the center panel (panel 3) had to be considered. Thus, a strong drive was required in order to overcome the superimposed resistive torques in consideration of the drive torque margin requirement. The high energy excess arising from this requirement under normal operating conditions would lead to

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a shock problem during latching. This could have been solved only by a complex and expensive viscous damping mechanism. The application of a simple stroke-dependent damper was not possible because latching of the antenna in a defined end position was required.

Those potential problems resulting from a spring drive led to the selection of a motor drive unit. The direct features are:

- Separation of the deployment steps (wing fixation during the deployment of the outer panel)

- Low deployment velocity and the possibility of a velocity-controlled deployment (providing low shock loads)

- Possibility of reverse operation in failure case

- Supply of high drive torque without any consequence to the shock load requirement

- High reduction gear (consequently the motor is less sensitive to the resistive torques of the antenna).

Selection of the motor type was guided by these considerations: Qualified drive units were available with or without brushes. Preference was initially given to a dc-brush motor without control equipment. The main disadvantage of a brush-less motor (stepper-motor) was seen in the need for control electronics which cause high costs and mass or space problems. However, after the first deployment tests at an advanced point of time, the deployment speed was found too high to meet the shock requirement. The deployment speed was then reduced and controlled by a small addition to the existing circuit hardware in the satellite system. But only one constant speed could be installed without any further reduction of speed at the end of deployment.

The dc-gearmotor is equipped with a redundant winding, a gearhead (reduction ratio 6000:1), and a torque limiter. The torque limiter consists of two friction discs with axial toothing. This slipping clutch was provided in order to protect the gearbox from high loads and to enable an overrunning capability of the motor in latched position.

The latching of the wing is performed by a two-bar system and a foldable element (see Fig. 7). During the deployment, the two-bar system is moved in the direction of its dead center configuration. In the end position, the two-bar system is locked by the stretched foldable element. The function of the foldable element is similar to the foldable bar of deployment Phase 1. Because of the known latching peak (see Fig. 6), the antenna will be accelerated at the end of deployment which causes higher shock loads. In order to balance the energy input from the foldable element, an additional damper was implemented. The design contained a small copper leaf (clamped

cantilever) which will be plastically deformed during the latching event of the antenna. Based on this improvement, the shock load requirement is met.

IN-ORBIT FUNCTION

The compliance with the stiffness requirement and the required accuracy of the deployed antenna is prerequisite for the in-orbit function.

The stiffness requirement was essentially met by provision of the DTS and its stiff locking mechanisms. In order to meet the accuracy requirement, several design features were introduced:

- Minimization of thermal deformations by selection of the CFRP-structural material

- Reduction of misalignment effects by

  - High quality demands to ball bearings and bearing fits

  - Application of titanium brackets at all hinge components (shafts and housings) in order to reduce thermal effects

  - Reproducible latching in defined end position (free of slipping).

Further improvements were gained from a shim procedure, in order to compensate actual misalignments due to manufacturing and integration tolerances. The antenna was aligned in a gravity compensation jig and the coordinates of 250 points on the deployed antenna were measured. A fine tuning of the panels' alignment was possible by shimming the interface points between panels and DTS, and by shimming the length of the foldable bars. The optimum shim corrections (location and thickness) were obtained from a computer program, based upon a finite element model of the antenna structure, plus optimization algorithm. This procedure was found very efficient, as no hardware iterations were needed, and the accuracy requirement was met after just one correction. The actual minimum/maximum deviations vertical to the frontside of the EM-antenna appeared in the range of ±0.8 mm at ground conditions.

CONCLUSIONS

Numerous mechanisms were needed for the hold-down, release, deployment, and locking of the SAR antenna. Due to its complexity, and the various sometimes conflicting requirements, straightforward designs had to be corrected, or improved, during the course of the development. Several component tests, and several analyses had to be performed to cover all essential details. Particular emphasis had to be given to the reliable performance of all those mechanisms as a failure could result in a complete loss of the satellite mission, rather than just a performance degradation.

Acceptable performance of all mechanisms was eventually verified at the first full antenna model.

ACKNOWLEDGMENTS

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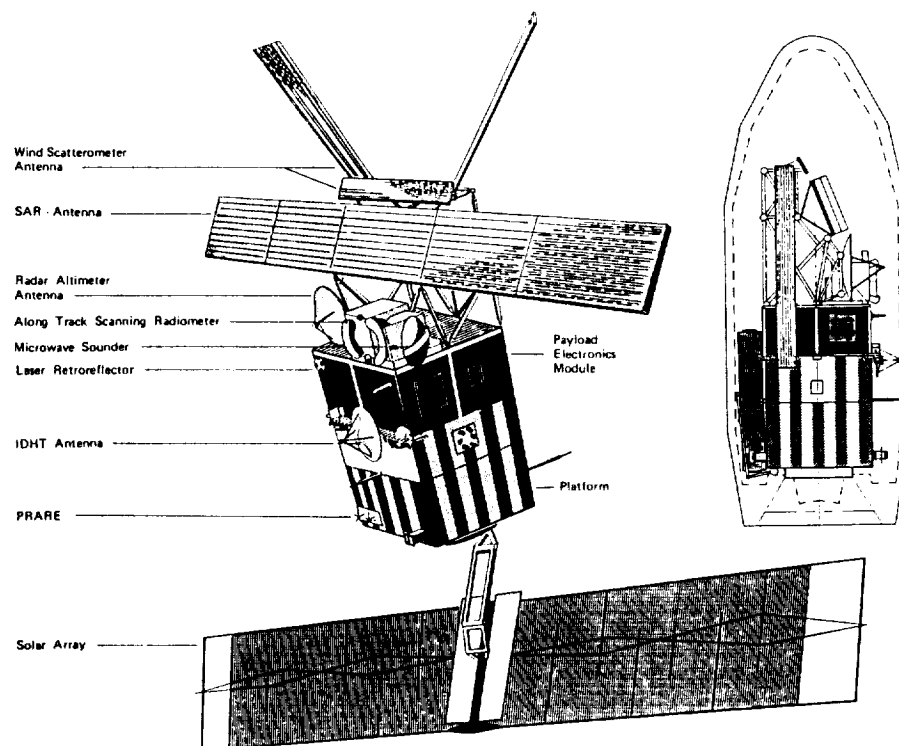
Wind Scatterometer Antenna

SAR · Antenna

Radar Altimeter Antenna

Along Track Scanning Radiometer

Microwave Sounder

Laser Retroreflector

IDHT Antenna

PRARE

Solar Array

Payload Electronics Module

Platform

Figure 1. ERS-1 satellite in launch- and fully-deployed configuration.

foldable bar

leaf spring

leg spring

leg spring

foldable bar

leaf spring

electric motor drive

locking rod

electric motor drive

locking rod
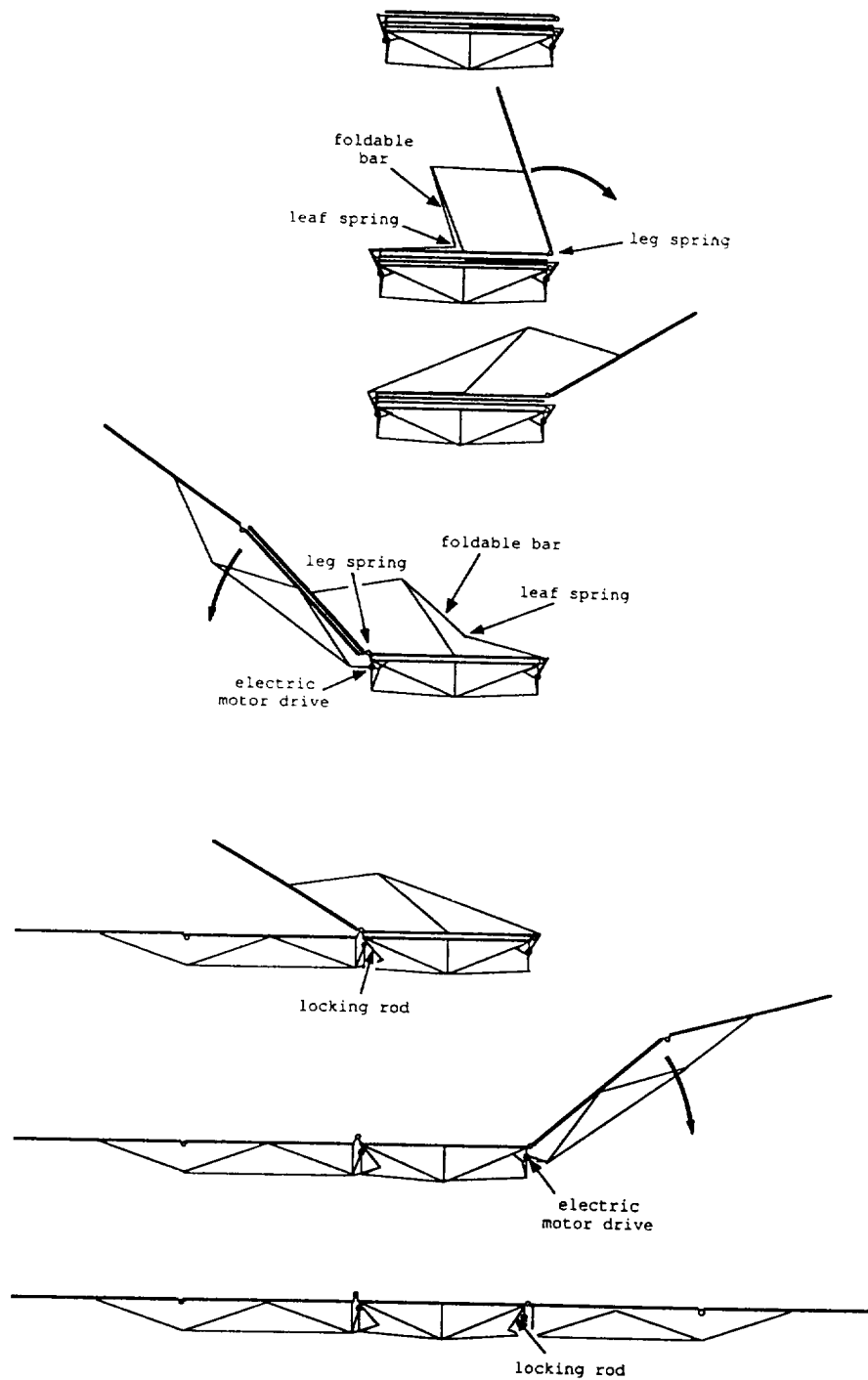
Figure 2. SAR-antenna deployment sequence.

LOCATION OF CLAMPING LEVER

LOCATION OF FIXATION PLATE

PANEL 1
PANEL 2
PANEL 5
PANEL 4
PANEL 3

y_A
z_A
x_A

Figure 3. SAR-antenna launch configuration.

4 - BAR LINKAGE LOCKING SYSTEM

TOGGLE
LEVER

RELEASE LEVER

PAWL

CLAMPING LEVER

HONEYCOMB SHOCK
ABSORBER

ARRESTING PIN

BRACKET PANEL 1

BRACKET
PANEL 2

BRACKET
PANEL 4

BRACKET
PANEL 5

BRACKET PANEL 3
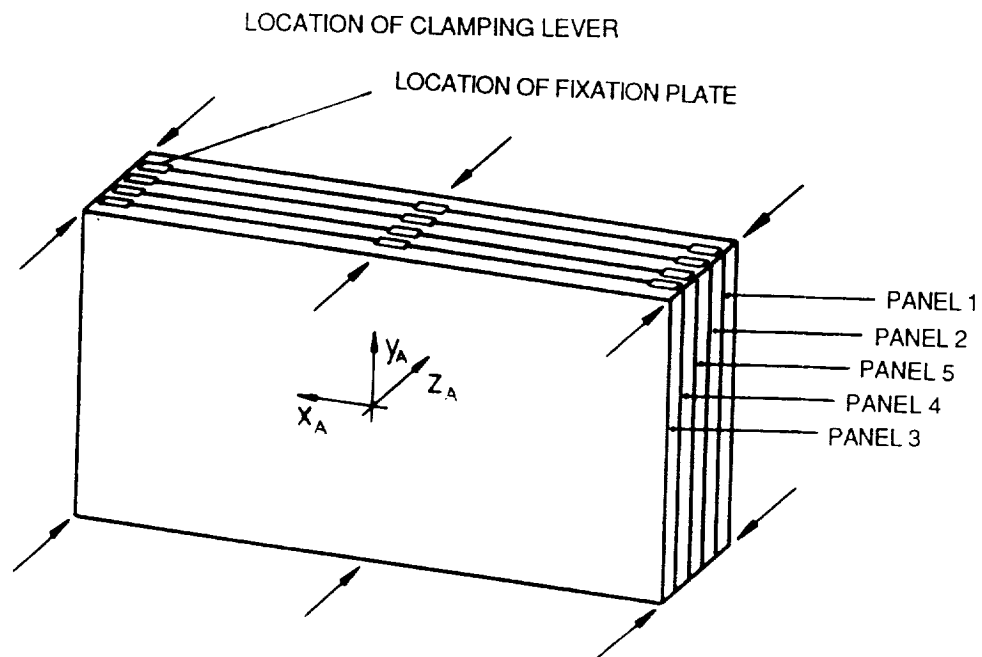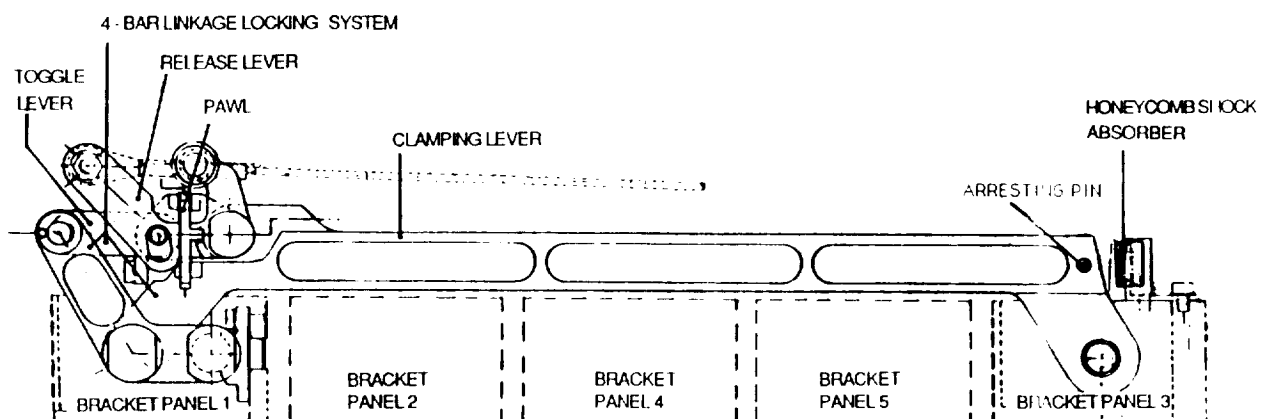
Figure 4. HRM clamping lever.
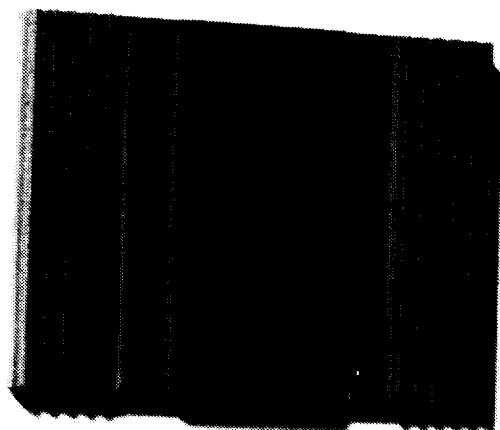
SINGLE FIXATION PLATE

FIXATION PLATES IN CONTACT

Figure 5.   Panel fixation plate.
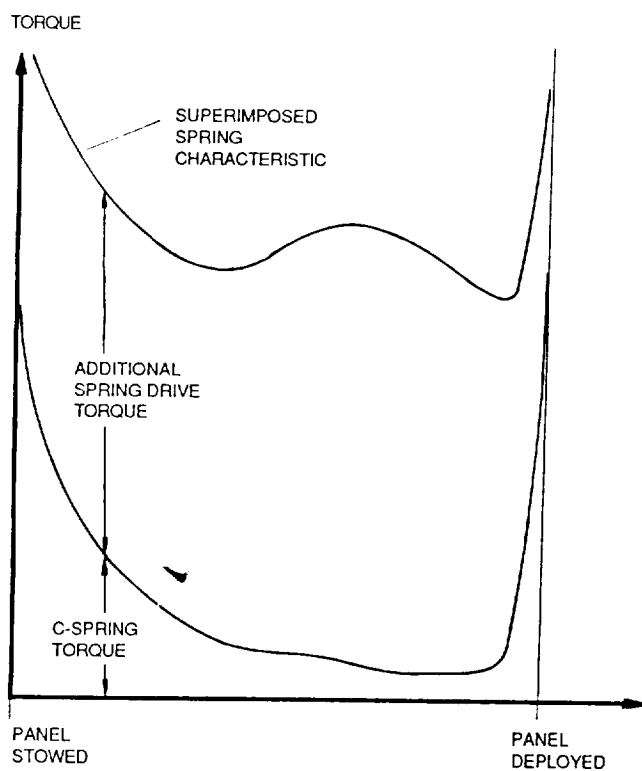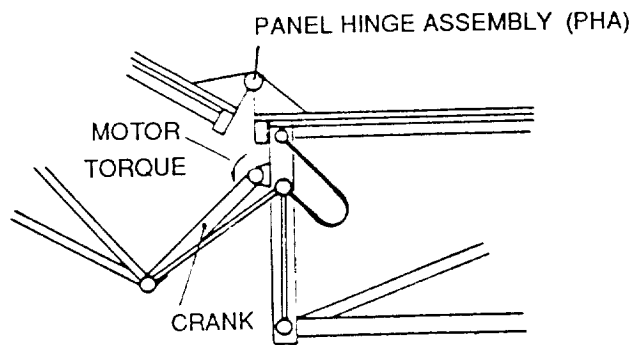
Figure 6.   Torque characteristic of phase 1 deployment (photo of
stretched leaf spring drive added).

INTERMEDIATE CONFIGURATION   DEPLOYED CONFIGURATION

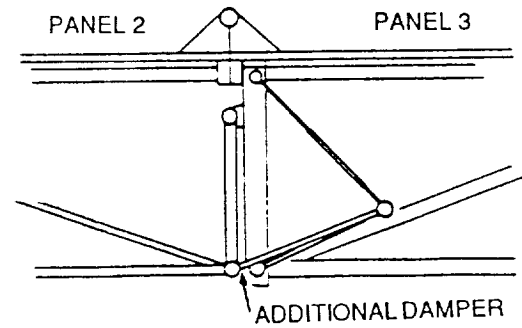PANEL HINGE ASSEMBLY (PHA)

MOTOR
TORQUE

CRANK

PANEL 2   PANEL 3

ADDITIONAL DAMPER

Figure 7.   Details of phase 2 latching.